

Vibration Characteristics of Industrial Gas Turbines



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According to *Power Engineering Magazine*, this year there have been over 200 announcements of new combustion or gas turbine installations in the United States. The announced projects will result in more than 66,000 MW of new capacity in the U.S. alone. As combustion turbines become a large component of the installed generation base worldwide, more engineers will be tasked with solving gas turbine vibration problems. The intent of this article is to introduce the vibration characteristics of large industrial gas turbines.

Three significant characteristics of large industrial gas turbines distinguish them from traditional turbomachinery.

1. **Gas turbine rotors are often constructed of several disks and distance pieces bolted together axially.**

These rotors are more flexible than traditional turbomachinery rotors. Although they are robust, behaviors commonly associated with such gas turbine rotors would be cause for concern if exhibited by typical turbomachinery.

This characteristic is compounded by the fact that combustion takes place in close physical proximity to the rotor.

During a cold startup of a typical large industrial gas turbine, the initial forty-five minutes to one hour of operation can show a significant increase or decrease in vibration. This phenomenon is typically called the thermal transient and is caused by the porting of compressor extraction air through the distance piece or marriage coupling into the turbine section for cooling air (Figure 1). The heat of combustion combined with the significantly cooler compressor extraction

causes distortion in the fits between rotor components. The distortion will result in binding and bowing of the rotor. The greater the tolerances of the fits between rotor components, the more significant the shaft bow. Manufacturers have implemented solutions to this problem such as the use of curvic couplings and welded transitions.

2. **Gas turbines typically operate above the second balance resonance.**

Most large industrial gas turbine rotors exhibit the first balance resonance between 1100 and 1600 rpm, the second balance resonance between 2400 and

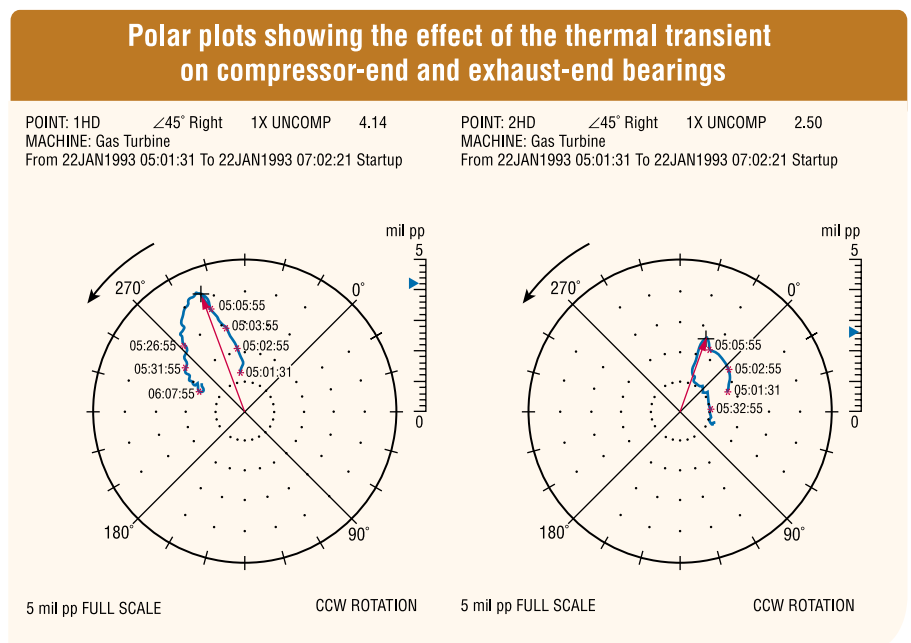


Figure 1.

Tangential bearing support for exhaust-end bearing

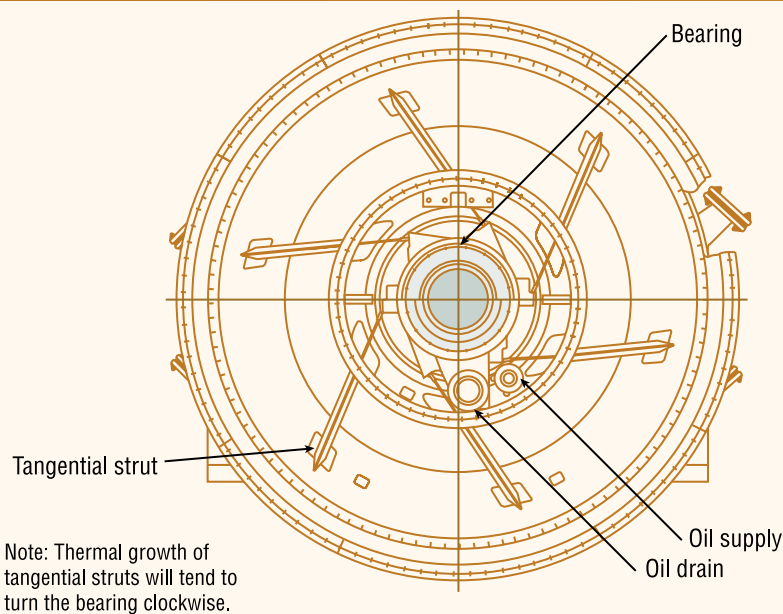


Figure 2.

2900 rpm. The third balance resonance is expected to be above running speed. These characteristic resonance speeds must be taken into consideration when analyzing vibration data for machinery faults and balancing. Because the running speed of the gas turbine is above the second mode, rather than between the first and second mode, this may invalidate commonly used two-plane balancing methods.

3. Gas turbine rotors are quite heavy in relation to the support structure.

Bearings at both the compressor end and the turbine or exhaust end are sometimes supported by radial or tangential struts within the gas path. The Dynamic Stiffness of the support structure may be significantly less than that of traditional turbomachinery. Casing interaction can be significant during transient (startup and shutdown) operation and during malfunctions. Dynamic Stiffness

of the compressor-end bearing can be significantly different from the Dynamic Stiffness of the exhaust-end bearing. Further complicating the machine, the Dynamic Stiffness in the horizontal direction is typically significantly less than in the vertical direction.

In addition to uncharacteristically compliant casings, high exhaust temperatures cause significant thermal growth and deformation of exhaust-end bearings. The different types of support structures for exhaust-end bearings have a significant effect on the apparent shaft average position and orbit shape (dynamic path of the shaft centerline).

As an example, Figure 2 shows a tangential bearing support for a gas turbine.

The thermal growth of the tangential supports will tend to turn the bearing clockwise (looking from the exhaust end), which is opposite the direction of rotation. This movement will distort the shaft centerline plot, at times making the shaft appear as though it is running outside of the bearing clearances. The shaft centerline plots in Figure 3 are typical shaft centerline plots from a common industrial gas turbine. These plots were compensated using startup data. Other models of gas turbines using radial and tangential bearing support structures will show similar characteristics.

Typical shaft centerline plot for a common industrial gas turbine

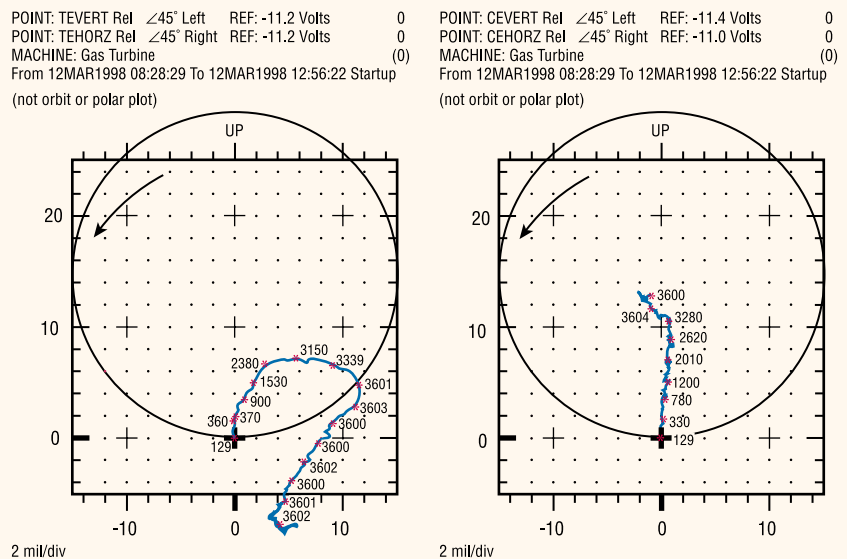
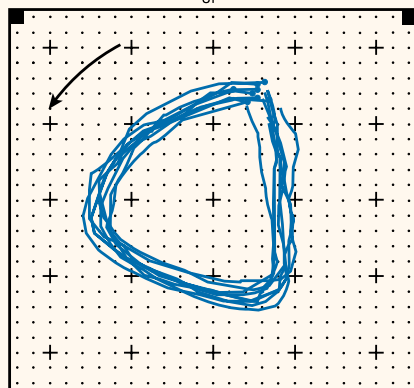


Figure 3.

Orbit plot from data taken during a previous startup (32 samples/rev)

Y: Bearing #2X Comp. $\angle 45^\circ$ Left DIR AMPL: 2.93 mil pp
 X: Bearing #2Y Comp. $\angle 45^\circ$ Right DIR AMPL: 3.01 mil pp
 MACHINE: GT-67
 15FEB1998 05:36:01 Shutdown DIRECT COMP UP



0.2 mil/div

ROTATION: X TO Y (CCW)

3598 rpm

Figure 4.

Conclusion

Gas turbines are highly complex machines. Flexible rotors, casings, compliant supports, and thermal extremes make them highly unusual pieces of rotating equipment. A recognition of the unique characteristics of gas turbines, along with a clear understanding of some of the basic concepts used in the analysis and characterization of traditional turbomachinery, should allow for a more fundamental understanding of behavior encountered in the field. The following case history is used to show the proper application of the basic concepts of machinery diagnostics. It also helps illustrate the unique characteristics of industrial gas turbines.

Case History

A large industrial gas turbine located in a process plant as part of a co-generation facility had a history of poor reliability since a rebuild three months previous. After a full load trip, it was decided that a closer look at the vibration data was necessary.

Upon arrival at the site, history and previous data were reviewed. Since the rebuild, the data indicated that any significant change in ambient temperature or unit load would result in high vibration and a subsequent trip. Because the nature of the problem was unknown by plant personnel, the unit had not been restarted after the last shutdown.

An orbit plot, Figure 4, from data taken during the most recent startup, one month earlier, showed indications of a

rub at the exhaust-end bearing. The rub was indicated by flatness on the right-hand side of the orbit. Although it seemed unusual that a rub could last for such a long period, the vibration data together with the symptoms exhibited by the machine seemed to support the theory.

An ADRE[®] for Windows[®] data acquisition instrument was connected to acquire vibration data during the startup. As a part of a continuing effort to balance the machine, several balance corrections had been attempted by plant personnel during the previous three months. Weights had been added to both the compressor and exhaust-end balance planes before the latest startup. Polar

plots using startup data showed that the unit passed through the first balance resonance at about 1200 rpm. The second balance resonance was expected to occur between 2600 and 2800 rpm. However, as the unit passed through this speed, none of the expected activity normally associated with a resonance occurred. At 3000 rpm, vibration increased rapidly to just above three mils peak-to-peak. At 3200 rpm, the increase in vibration suddenly stopped, and the phase changed 90° from 3200 to 3600 rpm. From this polar plot, it appeared that the second balance resonance was shifted from the 2600 to 2800 rpm range to just below running speed. Rotor-to-stator rubbing will increase direct support stiffness. An increase in stiffness will increase resonance speed. The pattern observed in Figure 5 is typical for a rub condition.

Figure 6 shows the same startup polar plot with orbit plots for several different speeds. Note the flat shape of the orbit plot indicating an abnormal path of the shaft centerline. In this

Polar plot of startup, exhaust-end bearing

POINT: TEHORZ Rel $\angle 45^\circ$ 1X UNCOMP 2.81 $\angle 26^\circ$ @3362 rpm
 MACHINE: Gas Turbine
 From 12MAR1998 08:28:29 To 12MAR1998 08:56:29 Startup

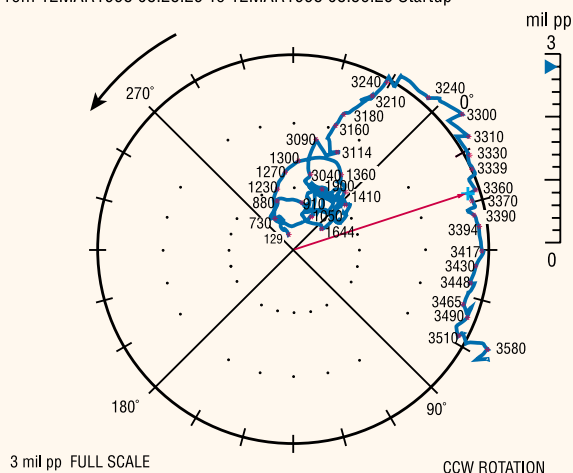


Figure 5.

Polar plot with shaft orbits at several speeds

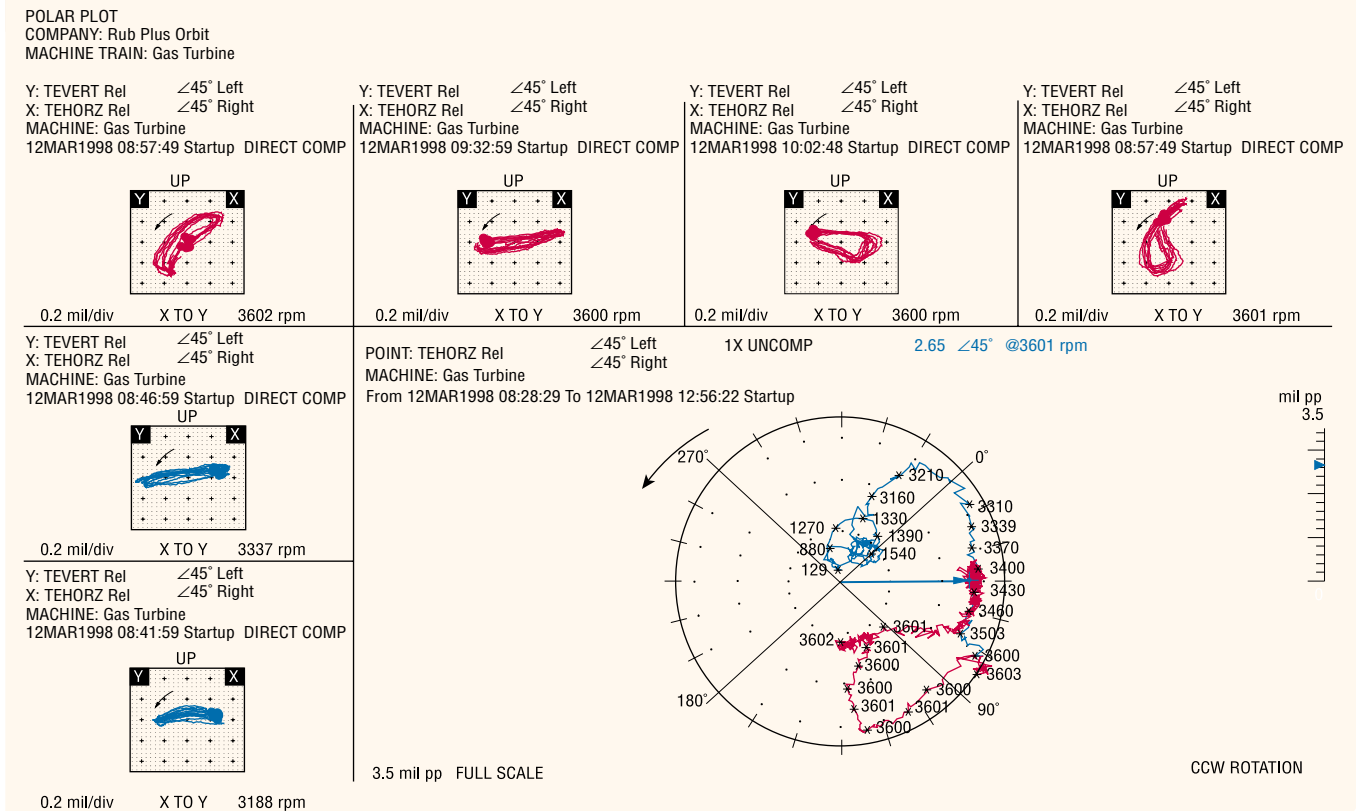


Figure 6.

case, the shaft appears to be heavily preloaded. As mentioned previously, a rub was suspected based on the shift in the 2nd balance resonance frequency with the unit coupled to its load. The presence of a heavy preload as evidenced by the orbit shape gave further support to a rotor-to-stator rub as the most likely malfunction.

As such, a check with the original equipment manufacturer (OEM) indicated that the clearances on turbine seals might have been reduced as a part of a recent upgrade.

The likelihood of reduced seal clearance and the strong evidence of rotor-to-stator rubbing from the

collected data subsequently led to disassembly of the machine and inspection of the seals. As expected, a seal rub was confirmed. Also, heavy deposits of oil and combustion by-products were found on the turbine seals. The seals were replaced and the unit was returned to service.

In conclusion, two elements of this case history are particularly noteworthy:

1. The deposits found on the seals resulted in a lubricated rub. Had this been a non-lubricated rub, the condition could not have persisted for three months without causing major damage to the machine. The ability to quickly

diagnose and pinpoint the location of a rub is very important.

2. An online system would have gathered the data automatically during the initial machine trip, three months previous. It also could have trended changes in amplitude and phase prior to the trip, possibly alerting plant personnel to a developing problem. In any event, an online system would have enabled more rapid identification and resolution of the problem rather than the costs incurred in re-starting the machine just to collect data.